PATENT SPECIFICATION

DRAWINGS ATTACHED

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COMPLETE SPECIFICATION

Improvements relating to Motor Vehicle Power Transmissions

We, HARRY FERGUSON RESEARCH LIMITED, of "Abbotswood," Stow-on-the-Wold, Gloucestershire, England, a British Company, do hereby declare the invention, for which we pray that a patent may be granted to us, and the method by which it is to be performed, to be particularly described in and by the following statement:—

This invention relates to motor vehicle power transmissions of the type comprising a hydromechanical variable-speed gear combination which incorporates an hydraulic torque converter including a casing, an impeller which is the input component and a turbine which is the output component.

It is notable in any such transmission that even at the lowest engine speeds, namely when idling, there is a drag through the relatively rotating converter components which causes an appreciable torque, so much so that the provision of a clutch in the transmission has been hitherto considered essential for a complete break between the engine and the road wheels or other driven parts of the vehicle.

It is also notable that the interposition of the hydraulic torque converter between the road wheels and the engine can at times deprive the vehicle of effective braking resistance from the engine. For instance, during braking, the normal brake system (namely the brakes applied to the road wheels or to their drive shafts) may get no effective assistance from the engine, unless the transmission includes a low gear and the driver changes to it.

The object of this invention is to provide a simple means, that becomes effective in a simple way, of attaining either of two purposes, namely:—nullifying drag torque through the hydraulic torque converter at low engine speeds; utilizing the converter to give braking assistance.

The invention is a motor vehicle power transmission of the type stated in which the hydraulic torque converter casing is connected to the engine to rotate with it, the turbine is connected to the converter-driven parts of the transmission and the impeller is connectible to but disconnectible from the casing, and in which rotation-restraining means under control of the driver are applicable to the impeller to restrain it against rotation.

An example of the hydromechanical variable-speed gear combination according to the invention is shown in the accompanying drawings, in which:—

Fig. 1 is a mid-section of the upper half of the hydraulic torque converter and associated gearing;

Fig. 2 is a perspective view of the hand lever, by operation of which the torque converter is controlled, and the gate by which the hand lever is guided and located;

Fig. 3 is a diagram of the hydraulic system in which the torque converter and the means for controlling it are incorporated, the system being shown as set for normal forward (or reverse) driving conditions;

Fig. 4 is a mid-section of a valve assembly also shown in Fig. 3, the valves being shown positioned to interpose a speed-reduction gear between the engine-driven casing and the impeller, and

Figs. 5 and 6 are views corresponding to Fig. 4 but showing the valves positioned to restrain the impeller against rotation.

The example is applied to power transmission apparatus substantially in accordance with the United Kingdom Patent Specification numbered 601,303 of Piero Mariano Salernie (now Count Giri de Teremala) in which apparatus a change-speed gear is interposed between the engine and the torque converter, the arrangement being such that either the engine and impeller rotate at the same speed or the engine rotates at a higher speed than the impeller, with consequent increase of torque.

Referring to Fig. 1 of the drawings, the

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casing of the hydraulic converter is indicated by 10, the vaned impeller by 11, the vaned turbine by 12 and the usual intervening reactor by 13. The casing is direct-coupled to the engine crankshaft so as to rotate always at engine speed. In the drawing, a projection 10A is partly shown on the casing; this is one of three equi-spaced connectors flexibly jointed by any conventional means to the engine crankshaft. The crankshaft is not shown in Fig. 1 but is indicated by 9 in Fig. 3.

The turbine 12 is keyed at 12A to the output shaft 14 of the converter. This is a hollow shaft which extends through a stationary gear case 15 enclosing the converter and other associated parts. The reactor 13 is supported on a stationary sleeve 16 through which the shaft extends. The reactor 13 is keyed at 13A to the one-way rotatable component 17 of the customary free-wheel device 18 by which the rotation of the reactor 13 is controlled.

The change-speed gear is interposed between the engine-driven casing 10 and the impeller 11. This gear is of simple epicyclic type, comprising a sunwheel 21, an internally toothed annulus 22, intervening planet wheels 23 and a planet carrier 25 with pins 24 on which these wheels are journalled. The planet carrier 25, which is the output member of the gear, comprises mainly spaced rings which are rigidly connected with the boss 11A of the impeller 11, so that the impeller and planet carrier form a single rotatable unit. The annulus 22 is the input member of the epicyclic gear, being secured to and forming an extension of the casing 10. The sunwheel 21 is the control member of the gear; it is formed as a ring of teeth on a sleeve shaft 26 which is journalled on bearings 26A on the reactor sleeve 16. This gear 21—25 may be regarded as an input epicyclic and may be called a "performance gear" seeing that it provides the driver of the vehicle with a means of accelerating the engine to work at a higher speed and produce greater power, or in other words give a better "performance."

In the example, the means for controlling the converter-epicyclic combination comprise three clutches 31, 32 and 33 each of the multiple friction-plate type.

The first clutch 31 is interposed between the impeller boss 11A and the sunwheel sleeve shaft 26. Thus, when this clutch is engaged—the second and third clutches 32 and 33 then being disengaged—the sunwheel 21 is locked to the impeller 11 and therefore to the planet carrier 25, the effect of which is that the epicyclic components are inter-locked and the engine is direct-coupled through the casing 10 to the impeller 11; that is to say, the transmission ratio is 1:1.

The second clutch 32 is interposed between the sunwheel sleeve shaft 26 and the stationary gear case 15 as hereinafter described. Thus, when the clutch 32 is engaged—the first and third clutches 31 and 33 then being disengaged—the sunwheel is held stationary, and so that impeller 11 is driven from the engine-driven casing 10 through the now effective epicyclic gear, the arrangement being such that the engine rotates at a higher speed than the impeller; in the example, the transmission ratio is 1.35:1.

The third clutch 33 is interposed between the planet carrier 25 and the stationary gear case 15, as hereinafter described. Thus, when the clutch 33 is engaged—the first and second clutches 31 and 32 then being disengaged—the impeller 11 (which forms part of the same rotatable unit as the planet carrier) is held against rotation.

The three clutches are hydraulically operated from a hydraulic system the pump of which is driven from and always rotates in unison with the engine crankshaft. In the example, the pump is of the external-internal gearwhel type; it is indicated by 34, 35. The inner gearwheel 34 of this pump is keyed at 34A to an extension 36, 36A of the annulus 22. The pump gearwheels are housed in a housing 37 which is a stationary fixture secured to the gear case 15.

The clutch 31 is under the control of an annular piston 38 in an annular "cylinder" 39 that is formed as an extension of the planet carrier 25. The piston 38 is arranged to act through presser pins 40 on the clutch 31 and being opposed by coil springs 41. The arrangement is such that the springs acts in the direction to engage the clutch, whereas the piston acts to disengage it. Hydraulic pressure fluid, namel oil, is led to the cylinder 39 by way of a conduit 42 through the reactor sleeve 16 and ports 42A in the sleeve 16 and sunwheel sleeve shaft 26. The fluid is supplied by the pump 34, 35 under the control of the valve assembly shown in Figs. 3, 4 and 5 hereinafter described.

The second and third clutches 32 and 33 also are under the control of the annular pistons, but in each instance the action is such that when hydraulic fluid is applied to the piston, the associated clutch 32 or 33 is engaged.

The piston of the clutch 32 is indicated by 43. This piston is fitted into an annular "cylinder" 44 formed in an internal body 45 which is a stationary fixture, being secured to the gear case 15. The cylinder 44 is supplied with hydraulic fluid from a conduit 46 and a port 46A, the conduit being under control of the already mentioned valve assembly.

The piston 47 of the clutch 33 is similarly fitted into an annular "cylinder" 48 also formed in the internal body 45. This third-clutch cylinder 48 is supplied with hydraulic fluid in the same way as the second-clutch cylinder 44, but the conduit and port are not shown in Fig. 1; they are indicated by 49 and 49A in Fig. 3.

It is to be understood that, in the normal forward (or reverse) drive of the vehicle, the

supply f hydraulic fluid is cut off from all three clutch cylinders 39, 44 and 48 and each is open to exhaust, namely to the sump in the transmission casing, from which sump the pump 34, 35 draws the fluid. Thus, the first clutch 31 is engaged under the pressure of its springs 41 whereas the other clutches are disengaged. Accordingly, as previously explained, the impeller 11 is free from restraint against rotation as a component of the hydraulic torque converter and the epicyclic gear components are inter-locked so that the transmission ratio between the engine and the impeller is 1:1.

In the example, the reversing gear of the vehicle also is a simple epicyclic gear comprising a sunwheel 51, annulus 52, planet wheels 53 and planet carrier 54. This gear may be regarded as an output epicyclic. The sunwheel is the input member of this output epicyclic; it is simply a ring of teeth on the converter output shaft 14. The annulus is the output member; it is a ring of internal teeth on a housing 55 keyed at 55A to the output shaft 56 of the total hydromechanical variable-speed gear combination shown in the drawing. The output epicyclic is under the control of a gear shift hand lever, not shown in Fig. 1, but which is indicated by 60 in Fig. 2 and which is devised to operate in any conventional manner on the customary fork 60A secured to a longitudinally slidable rod 60B. The fork 60A engages with an annular groove 62 in a collar 63 which is formed as a doubly toothed dog clutch. This clutch is slidably splined at 67 on the boss 61 of the planet carrier 54; the clutch is shown in its neutral position in Fig. 1. When the clutch is positioned to give forward drive, its outer teeth 64 engage the annulus teeth 52. When the clutch is positioned to reverse the drive, its inner spline teeth 65 engage teeth 66 on a stationary fixture 68 secured to the gear case 15.

The already mentioned gear shift hand lever 60 of the output epicyclic 51—55 is interconnected with the valve assembly according to Figs. 3 to 6 by operation of which the hydraulic system is caused to brake the impeller 11 and disable the input epicyclic 21—25 as a drive-transmitting means or to free the impeller and re-establish the input epicyclic, as the case may be. The arrangement is such that when a gear shift has to be effected, either from forward to reverse or *vice versa*, the vehicle being stationary, the action causes temporary restraint of the impeller 11 against rotation. Thus, the impeller nullifies the drag torque ordinarily transmitted through an idling torque converter, and so the driver of the vehicle is enable to perform the gear shift without need for disengagement of a separate friction clutch, no such clutch being provided.

The hand lever 60 controlling the output epicyclic is arranged to be shifted in a gate 70, shown in Fig. 2, providing a guideway 71 with three spaced notches 72, 73 and 74 respectively defining the lever positions for "forward," "neutral" and "reverse." The guideway also has a fourth spaced notch 75 defining the position for utilising the impeller to assist in braking the vehicle; this notch may be termed the "emergency braking" position. It will be obvious that the hand lever when manipulated can perform to-and-fro movements in two directions, namely lengthwise of the guideway 71 and transversely of the guideway when the lever registers with any of the notches. When the lever 60 is moved lengthwise of the guideway, it is effective to move the slide rod 60B, fork 60A and doubly toothed clutch 63. The transverse movements of the lever into and out of the notches serves another purpose, namely, to displace a spring-returned valve hereinafter described which may be called the "impeller brake valve" and the stem of which is indicated by 76 in Fig. 2.

Referring more particularly to Fig. 2 the foot of the hand lever 60 is a yoke 80 which is connected by a pivot pin 81 to the down-turned end of a transverse rod 82 with a finger 83 which engages a slotted projection 84 on the fork 60A. The rod 82 extends freely into the yoke 80. The yoke is fitted with an abutment 85. The hand-lever assembly includes a lever 86, 87 which is supported by a vertical-axis pivot pin 88 on a bracket 89 which is a fixture on the gear case 15. The arms 86, 87 of this lever bear respectively against the abutment 85 and valve stem 76, which latter is continuously urged against the arm 87 by the action of the return spring of the valve. The operation is as follows:—

When the hand lever 60 is moved transversely of the gate from or into a notch, it pivots about the pin 81 and pivots the lever 87 so that the valve stem 76 is moved in one direction or returns in the opposite direction; when the hand lever is moved along the guideway 71 it pivots around the axis of the rod 82, turning this rod and its finger 83 so that the clutch 63 is shifted.

In the example shown, the emergency braking notch 75 has a projection 90 which safeguards the driver against inadvertent entry of the hand lever 60 into this notch.

Moreover, the hand lever has a small slidable sleeve 91 which neatly fits the notches 72 to 75. This sleeve is connected to a small internal rod 92, the top of which is a push button 93. The sleeve 91 is maintained in its normal position by a spring 94, in which position it cannot pass the projection 90. Thus, if the driver wishes to brake the impeller 11, he must depress the button 93, in which event the hand lever 60 can be put into the notch 75, following which the button is released and the sleeve 91 slides upwards into this notch.

With reference now to the valve assembly

according to Figs. 3 to 6, in which the impeller brake valve is indicated by 100, for convenience of description it will be assumed that the valve is pushed *inwards* by the hand lever 60 and is returned *outwards* by its spring 101. It will be seen that the valve 100 can occupy any one of three positions, namely:—an outer position (Figs. 3 and 4) when the hand lever is offset transversely to occupy any of the "forward," "neutral" and "reverse" notches 72, 73 and 74; an inner position (Fig. 5) when the hand lever 60 is in the guideway 71; an innermost position (Fig. 6) when the hand lever 60 is in the emergency braking notch 75. The inner and innermost positions of the valve 100 have functionally the same effect.

With reference to Fig. 3, the valve assembly shown diagrammatically by way of illustration includes a stationary valve chest 102, which is formed with a main vertical bore 103 containing the valve 100 and return spring 101. The chest is formed with another bore 104 containing a valve 105; this valve may be called the "performance gear valve," as hereinafter explained. This performance gear valve 105 can be conveniently moved by utilising the low voltage electric system of the vehicle. Thus, the valve can be connected to a switch (for example on the steering column) controlling an electric circuit containing a solenoid connected to the valve and operative against a return spring. The arrangement is such that when the switch is put "on" the valve is pulled inwards into its Fig. 4 or Fig. 6 position, independently of the position of the hand lever 60 and valve 100, and when the switch is put "off" the valve is returned outwards into its Fig. 3 or Fig. 5 position.

The valve chest 102 has the following conduits connected to it, namely:—an inlet conduit 106 for hydraulic pressure fluid from the pump 34, 35, which draws from the sump three conduits 111, 112, 113 for the passage of the hydraulic fluid to and from the already mentioned conduits 42, 46 and 49, respectively, of the three clutches 31, 32 and 33. The valve chest 102 has an hydraulic pressure system of ports 115 with which the inlet 106 connects. The valve chest 102 also has an exhaust system of outlet ports 114 through any of which the fluid can exhaust to the sump. The conduits 111, 112, and 113 respectively connect with port system 116, 117 and 118 within the valve chest. As shown, the valve 100 has ports 120, 121, 122 and 123 and the valve 105 has ports 124 and 125, all of which ports co-operate with the port system in the chest.

The valve assembly, according to Fig. 3, is set to connect all three clutch cylinders 39, 44 and 48 to exhaust. That is to say, the three conduits 111, 112 and 113 are respectively connected through the ports 124 and 120, the port 125 and the port 123 with the exhaust system 114. Thus, the first clutch 31 is engaged and the second and third clutches 32 and 33 are both disengaged; and therefore the sunwheel 21 is locked to the planet carrier 25 and the drive (either forward or reverse) is transmitted from the engine through the interlocked epicyclic components 21, 25 to the impeller 11 with the ratio of 1:1.

Referring to Fig. 4, the valve 105 is moved into its inner position, in which the cylinders 39 and 44 of the first and second clutches become connected through the port 124 and the ports 125 and 121 with the hydraulic pressure port system 115 which is connected by the conduit 106 with the pump. The cylinder 48 of the third clutch remains connected by the port 123 of the valve 100 to the exhaust port system 114. Accordingly, the first clutch 31 becomes disengaged and the second clutch engaged, and so the sunwheel 21 is freed from the planet carrier 25 but is locked to the gear case 15. Thus, the epicyclic now performs its true epicyclic function and transmits the drive from the engine to the impeller at the selected ratio such that the engine speed is higher than the impeller speed. It is because of the foregoing effect that the valve 105 may be regarded as bringing the "performance gear" into action and therefore may be called the "performance gear valve."

Referring to Fig. 5, the valve 100 is shown in its inner position, the performance gear beng out of operation; i.e. the valve 105 is in its outer position. As Fig. 5 shows, the first and third clutch cylinders 39 and 48 are connected through the ports 124 and 120 and the port 123 to the hydraulic pressure port system 115 whereas the second clutch cylinder 44 is connected through the port 125 to the exhaust port system 114. Accordingly, the first and second clutches 31 and 32 are both disengaged, so that the sunwheel 21 is freed both from the planet carrier 25 and the gear case 15, but the third clutch 33 tends to lock the planet carrier 25 and with it the impeller 11 to the gear case and therefore restrains the impeller against rotation; and so the valve 100 may be called the "impeller brake valve."

It will be apparent that the driver of the vehicle, during normal driving at 1:1 ratio (Fig. 3), may engage the performance gear (Fig. 4) and reduce the engine speed so as to apply a light brake action to the vehicle. Thereafter, he may apply the full brake effect derived from locking the impeller by putting the hand lever into the notch 75. In this event, the valves 100 and 105 occupy the positions shown in Fig. 6, in which the valve assembly performs the same functions as in Fig. 5 position. In this instance, however, the first clutch cylinder 39 is connected through only the valve 105, by way of the port 124, to the pressure port system 115, and the second clutch cylinder 44 is connected through both valves 105 and 100, by way of the ports 125

and 122, to the exhaust port system 114.

Figs. 5 and 6 show the impeller brake valve 100 in the inner and innermost positions which it occupies when the hand lever 60 is in the guideway 71 and the impeller braking notch 75, respectively. These positions both have the same effect, this being achieved by making the valve ports 120 and 123 wide enough to register with the pressure system ports 115 in both positions.

It will be manifest that operation of the impeller brake valve 100 serves the two functions previously referred to namely:—(1) while the vehicle is at rest with the engine idling, the braked impeller prevents the transmission of torque from the revolving converter casing 10 to the turbine 12, so that change from forward to reverse or *vice versa* through the doubly toothed clutch 63 may be performed; (2) while the engine is travelling forwards (or in reverse) under normal conditions (namely 1:1 ratio through the epicyclic gear), the braked impeller serves as a means of deriving braking effort from the converter. Thus, when driving down a steep hill the driver needs simply shift the hand lever 60 across the guideway 71 into the "emergency braking" notch 75, in which event the impeller 11 will be restrained against rotation, whereas the casing 10 continues to rotate with the engine and the turbine 12 is driven by the road wheels. On the one hand the turbine and to a lesser extent the casing will tend to circulate the oil in the converter, whereas on the other hand the stationary impeller will nullify this tendency, with the result that the converter will function as a brake.

The hydromechanical variable-speed gear combination constitutes a complete continuously variable change speed gear for transmitting driving torque from the engine to the propeller shafting of the vehicle; that is to say, there is no need for a gear box, at least for vehicles of the various ordinary "private car" types. Nevertheless, where the combination is applied to a heavy duty vehicle, such as for instance a heavy truck, the output shaft 56 may be connected to a gear-box.

WHAT WE CLAIM IS:—

1. A motor vehicle power transmission of the type stated in which the hydraulic torque converter casing is connected to the engine to rotate with it, the turbine is connected to the converter-driven parts of the transmission and the impeller is connectible to but disconnectible from the casing, and in which rotation-restraining means under control of the driver are applicable to the impeller to restrain it against rotation.

2. A motor vehicle power transmission of the type stated including also a speed reduction epicyclic gear between the engine and the hydraulic torque converter, in which the converter casing is connected to the engine to rotate with it, the turbine is connected to the converter-driven parts of the transmission and the impeller is connected to the output member of the epicyclic gear so as to be driven thereby at a selected ratio and is connectible to a stationary part so as to be restrained against rotation.

3. A motor vehicle power transmission of the type stated including also a speed reduction epicyclic gear between the engine and the casing of the hydraulic torque converter, in which the converter casing is connected to the annulus of the epicyclic gear, the impeller is rotatable in unison with the planet carrier of the gear and the turbine is rotatable in unison with the converter-driven parts of the transmission, and in which the planet carrier and impeller are connectible to the sunwheel of the gear or alternatively the sunwheel is connected to a stationary part to restrain it against rotation or alternatively the planet carrier and impeller are connectible to siad stationary part to restrain them against rotation.

4. A motor vehicle power transmission according to claim 1, 2 or 3 in which a changeable gear is interposed between the hydraulic torque converter and the road wheels of the vehicle and in which driver-operated means for effecting change of said gear is connected to means for causing rotation-restraint of the impeller so that a gear change can be effected only while the impeller is restrained against rotation.

5. A motor vehicle power transmission according to claim 4 in which the changeable gear is a reversing gear.

6. A motor vehicle power transmission according to claim 5 in which the reversing gear is of the epicyclic type, the sunwheel being the input member and the annulus the output member, and includes a toothed clutch for connecting the planet-carrier either to the annulus or to a stationary part.

7. A motor vehicle power transmission according to any of claims 4, 5 and 6 in which the changeable gear is under the control of a hand lever that is movable in a gate formed with a guideway, lever-positioning notches opening sidewise from the guideway corresponding to different gear settings, said lever being connected to a gear-changing component so as to move it when the lever is moved along the guideway and said lever being also connected to means for restraining rotation of the impeller so as to operate said means when the lever is moved out of or into any of the notches.

8. A motor vehicle power transmission according to claim 7 in which the gate has forward, neutral and reverse notches at one side of the guideway and a lever-positioning notch at the opposite side corresponding to rotation-restraint of the impeller.

9. A motor vehicle power transmission according to any of claims 3 to 8 in which

the connection between the planet-carrier and impeller and the sunwheel, the connection between the sunwheel and a stationary part and the connection between the planet-carrier and impeller and said stationary part are operable by clutches under the control of an impeller brake valve and a performance gear valve, said valves being incorporated in a valve assembly interposed between a pressure fluid system and hydraulic devices applied to the respective clutches.

10. A motor vehicle power transmission substantially as hereinbefore described with reference to and as shown in Figs. 1 and 2 of the accompanying drawings.

11. A motor vehicle power transmission substantially as hereinbefore described with reference to the accompanying diagrammatic drawings Figs. 3 to 6.

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PROVISIONAL SPECIFICATION

Improvements relating to Motor Vehicle Power Transmissions

We, HARRY FERGUSON RESEARCH LIMITED, a British Company, of "Abbotswood", Stow-on-the-Wold, Gloucestershire, England, do hereby declare this invention to be described in the following statement:—

This invention relates to motor vehicle power transmissions of the type comprising a hydromechanical variable-speed gear combination which incorporates an hydraulic torque converter consisting of a casing, an impeller which is the input component, a turbine which is the output component and usually an intervening reactor.

It is notable in any such transmission that even at the lowest engine speeds, namely when idling, there is a drag through the relatively rotating converter components which causes an appreciable torque, so much so that the provision of a clutch in the transmission has been hitherto considered essential for a complete break between the engine and the road wheels or other driven parts of the vehicle.

It is also notable that the interposition of the hydraulic torque converter between the road wheels and the engine can at times deprive the vehicle of effective braking resistance from the engine. For instance, during braking, the normal brake system (namely the brakes applied to the road wheels or their drive shafting) may get no effective assistance from the engine, unless there is a low gear and the driver changes to it.

The object of this invention is to provide a simple means, that becomes effective in a simple way, of nullifying drag torque through the hydraulic torque converter at low engine speeds and of utilizing the converter to give braking assistance.

The invention is a motor vehicle power transmission of the type stated in which the converter casing is connected to the engine to rotate with it, the converter turbine is connected to the converter-driven parts of the transmission and the converter impeller is connectible to but disconnectible from the engine, and in which rotation-restraining means under control of the driver are applicable to the impeller to restrain it against rotation.

An example of the hydromechanical variable-speed gear combination according to the invention is shown in the accompanying drawing, which is a mid-section of the torque converter and associated gearing.

The example is applied to power transmission apparatus substantially in accordance with the United Kingdom Patent Specification numbered 601,303 of Piero Mariano Salerni (now Count Giri de Teramala) in which apparatus a reduction gear is interposed between the engine and the torque converter to drive the impeller either at engine speed or at a slower speed with consequent increase of torque.

Referring to the drawing, the casing of the hydraulic converter is indicated by 10, the vaned impeller by 11, the vaned turbine by 12 and the intervening reactor by 13. The casing is direct-coupled to the engine crankshaft so as to rotate always at engine speed. The turbine is keyed to the output shaft 14 of the converter. This shaft extends through a stationary gear case 15 associated with the converter. The reactor 13 is keyed to a sleeve shaft 16 through which the shaft 14 extends, there being appropriate bearings between the two shafts. The reactor shaft 16 has keyed to it the one-way rotatable component 17 of the customary free-wheel device by which the rotation of the reactor is controlled. The other component of this device is a ring 18 which is a fixture within the gear case 15.

The reduction gear is interposed between the engine-driven casing 10 and the impeller 11. This gear is of simple epicyclic type, comprising a sunwheel 21, an internally toothed annulus 22, intervening planet wheels 23 and a planet carrier with pins 24 on which these wheels are journalled. The planet carrier comprises mainly the boss 11A of the impeller and it includes also a supporting ring 25. The annulus 22 is the input member, being secured to and forming an extension of the casing 10. The planet carrier is the output member of the epicyclic gear, although it forms part of the same unit as the impeller of the converter. The sunwheel 21 is the control member of the gear; it is formed as a ring of teeth on a sleeve shaft 26 which is journalled on bearings on the reactor shaft 16. This gear 21—25 may be regarded as an input epicyclic.

In the example, the means for controlling the sunwheel of the input epicyclic comprise a multiple friction-plate clutch 31 and separate brake bands 32 and 33. The clutch is interposed between inner and outer portions of two co-axial drums 34 and 35 which are arranged side by side and which have brake rims 36 and 37 respectively embraced by the bands 32 and 33. The drum 34 is keyed to the sunwheel shaft 26, whereas the other drum 35 is keyed to a sleeve 38 which is rotatable on this shaft 26 and which forms a rigid extension of the impeller-and-planet-carrier unit.

The clutch 31 and brake bands 32 and 33 are hydraulically operated from a hydraulic system the pump of which is driven from and always rotates in unison with the engine crankshaft. In the example, the pump is of the external-internal gearwheel type; it is

indicated by 41. The inner gearwheel of this pump is keyed to a tubular extension 42 of the annulus 22. The clutch 31 is under the control of an annular piston 43 in a cylinder 44 that is formed in the drum 35, this piston being arranged to act through a presser 45 on the clutch 31 and being opposed to a "Belville" spring 46. The arrangement is such that the spring acts in the direction to engage the clutch, whereas the piston acts to disengage it. Each of the brake bands 32 and 33 is acted upon by a small hydraulic band-tightening ram (not shown) in the customary way, each of the rams being mounted upon the gear case 15. The brake band 33 is normally disengaged; that is to say, normally no restraint is put on the rotation of the impeller 11. The operation of the band 33 will be described later herein. The clutch 31 and brake band 32 are under the control of a two-position tap (not shown), which is the means whereby the driver switches the reduction gear in or out. In one position of the tap, the clutch 31 is engaged and the brake band 32 disengaged. Thus, the sunwheel 21 is locked to the planet carrier so the engine is direct-coupled to the impeller 11; *i.e.*, the transmission ratio is 1:1. In the other position of the tap, the brake band 32 is engaged and the clutch 31 disengaged. Thus, the sunwheel is held stationary and so the impeller is driven at reduced speed; in the example, the reduction ratio is 1.39:1.

In the example, the reversing gear of the vehicle also is a simple epicyclic gear comprising a sunwheel 51, annulus 52, planet wheels 53 and planet carrier 54. This gear may be regarded as an output epicyclic. The sunwheel is the input member of this output epicyclic; it is simply a ring of teeth on the converter output shaft 14. The annulus is the output member; it is a ring of internal teeth on a housing 55 provided on the output shaft 56 of the total hydromechanical variable-speed gear combination shown in the drawing. The output epicyclic is under the control of a gear shift hand lever (not shown) which operates an axially shiftable rod 61 that has a fork 62 in engagement with a doubly toothed clutch 63. This clutch is slidably splined at 67 on the planet carrier 54; it is shown in its neutral position. When the clutch is positioned to give forward drive, its teeth 64 engage the annulus teeth 52. When the clutch is positioned to reverse the drive, its teeth 65 engage teeth 66 on the fixture 18.

The hand lever of the output epicyclic is inter-connected with a valve gear by operation of which the hydraulic system is caused to brake the impeller and disable the input epicyclic or to free the impeller and re-establish the input epicyclic, as the case may be. The arrangement is such that when a gear shift has to be effected, either from forward to reverse or *vice versa*, the action causes temporary restraint of the impeller 11 against rotation. Thus, the impeller nullifies the drag torque ordinarily transmitted through an idling torque converter, and so the driver is enabled to perform the gear shift as a self-sufficient operation without need for disengagement of a separate friction clutch.

The hand lever controlling the output epicyclic is arranged to be shifted in a gate providing a guideway with three spaced notches defining the lever positions for forward, neutral and reverse. The arrangement is such that under normal conditions, when the hand lever occupies any one of the notches, the impeller brake band 33 is disengaged. Whenever the lever is moved into the guideway from any one of the three notches, the valve gear is operated so that it performs two simultaneous actions, namely: disables the input epicyclic by disengaging the clutch 31 or brake band 32, whichever was engaged; restrains the impeller 11 against rotation by engaging the brake band 33. Thus, the stationary impeller 11 in the idly revolving casing 10 will prevent the transmission of drag torque to the turbine 12. This condition continues only so long as the hand lever occupies the guideway. Thus, the shift of the doubly toothed clutch 63 is completed before normal conditions are restored by the return movement of the lever into one of the notches.

It will be manifest that the hand lever provides a simple means of utilizing the hydraulic torque converter as a means of deriving braking effort from the engine. Thus, when driving down a steep hill the driver need simply shift the hand lever into the guideway into a marked position at or near the "forward" notch, in which the clutch teeth 64 and 52 are still inter-engaged. In this event the impeller will be restrained against rotation, whereas the casing continues to rotate with the engine and the turbine is driven by the road wheels. On the one hand the turbine and to a lesser extent the casing will tend to circulate the oil in the converter, whereas on the other hand the stationary impeller will nullify this tendency, with the result that the converter will function as a brake.

The hydromechanical variable-speed gear combination shown in the drawing constitutes a complete continuously variable change speed gear for transmitting driving torque from the engine to the propeller shafting of the vehicle; that is to say, there is no need for a gear box, at least for vehicles of the various ordinary "private car" types. Nevertheless, where the combination is applied to a heavy duty vehicle, such for instance as a heavy truck, the output shaft 56 may be connected to a gear-box.

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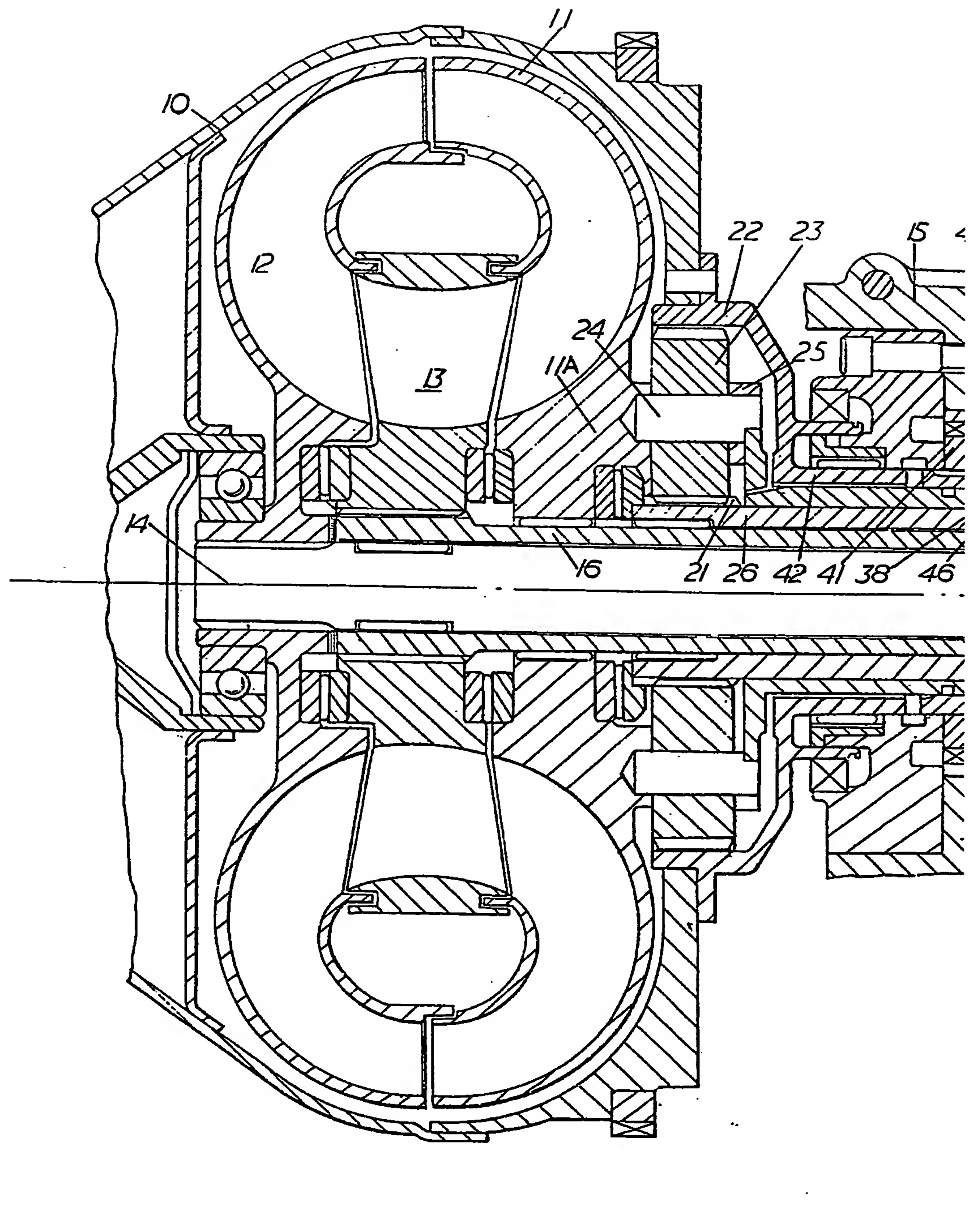

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870,863 **PROVISIONAL SPECIFICATION**

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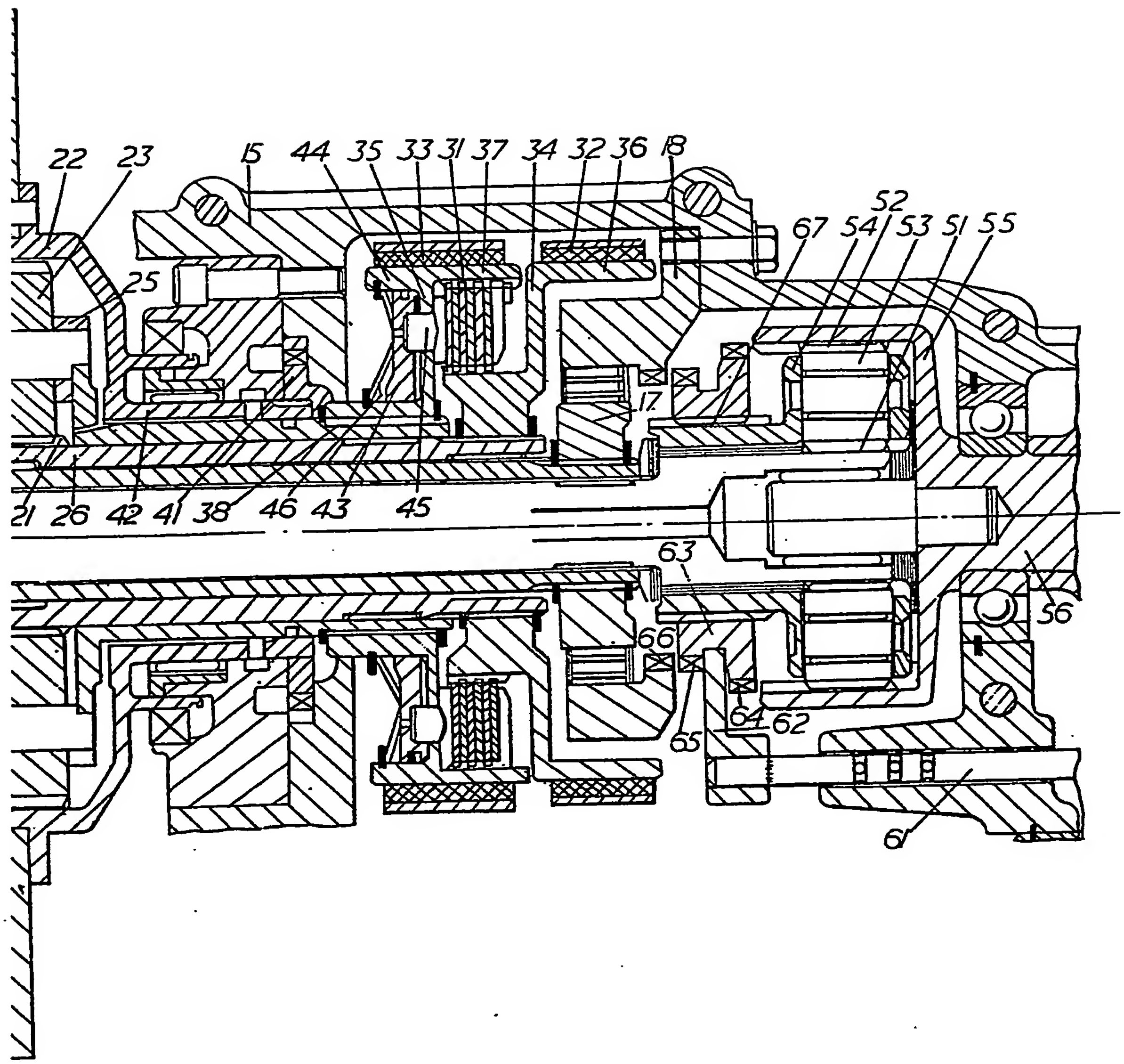

870,863 PROVISIONAL SPECIFICATION

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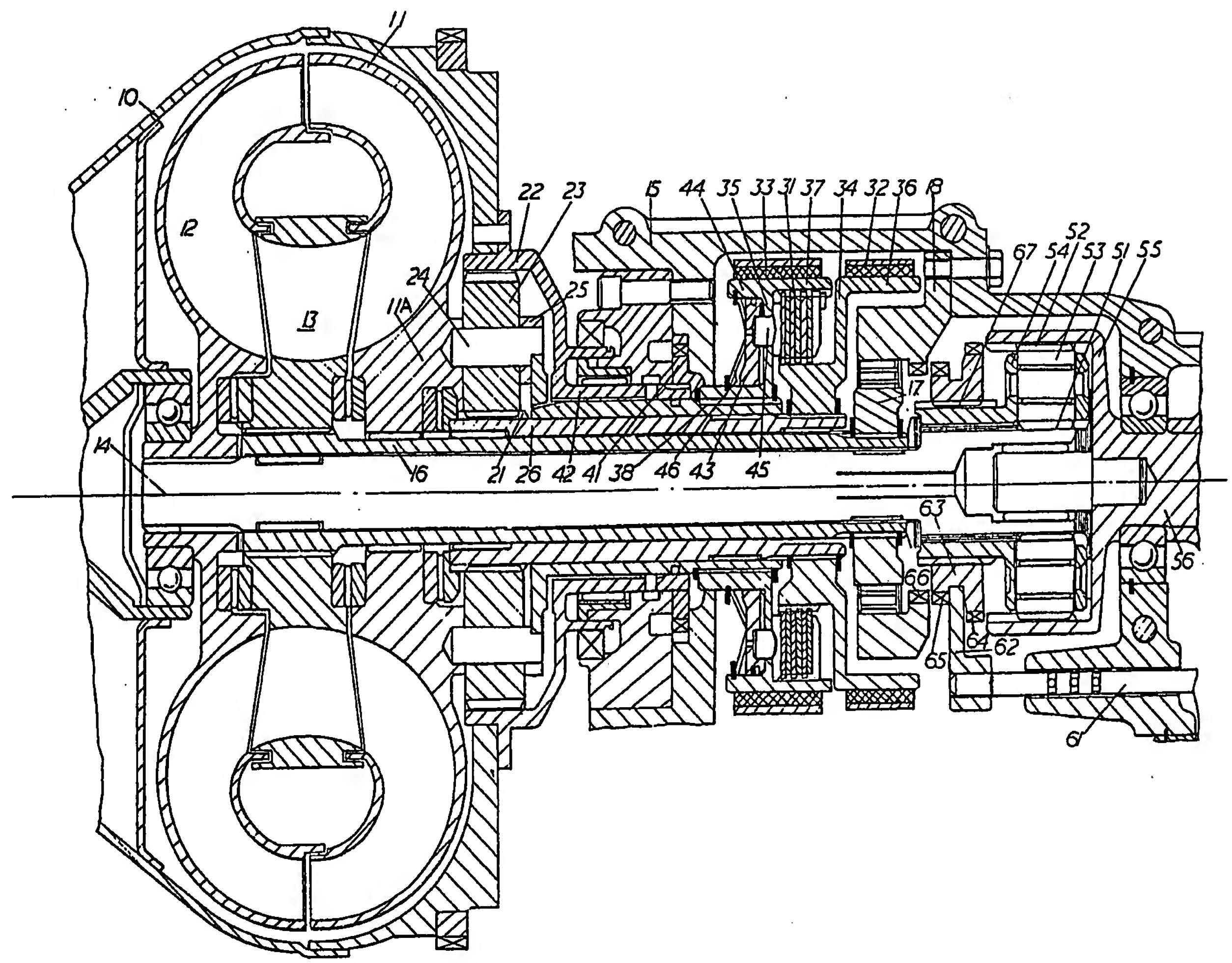

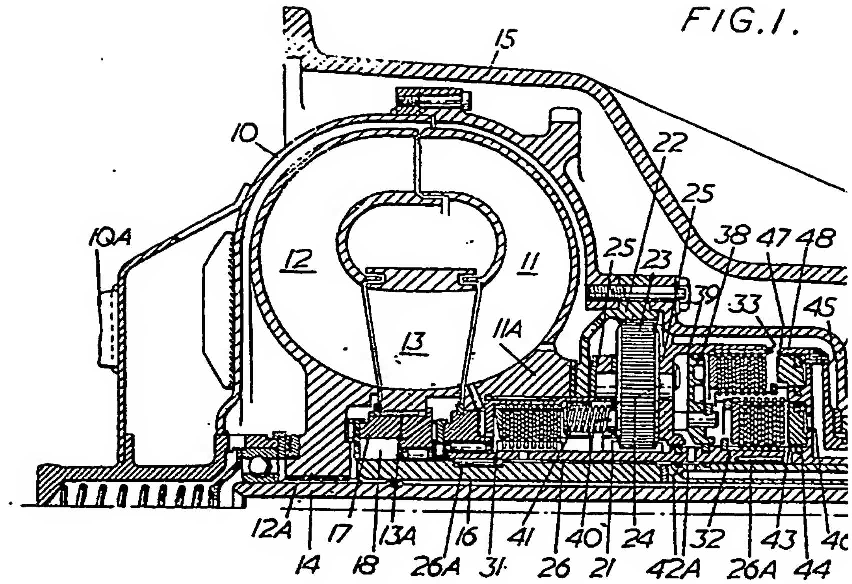
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FIG. 1.

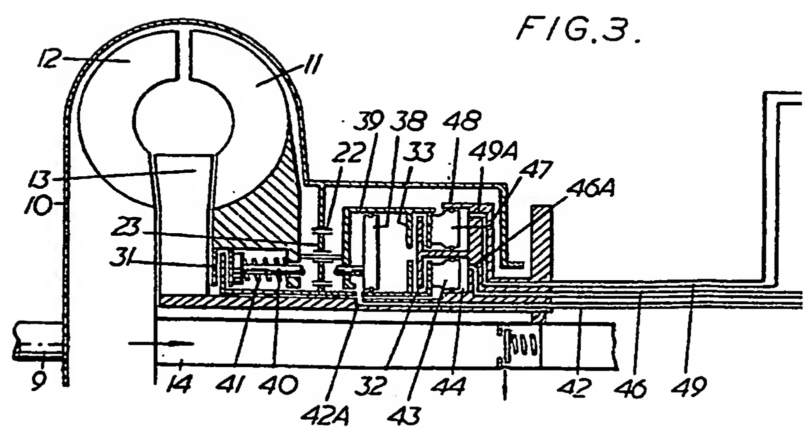
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FIG. 3.

870,863 COMPLETE SPECIFICATION
3 SHEETS
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SHEET 1

FIG. 1.

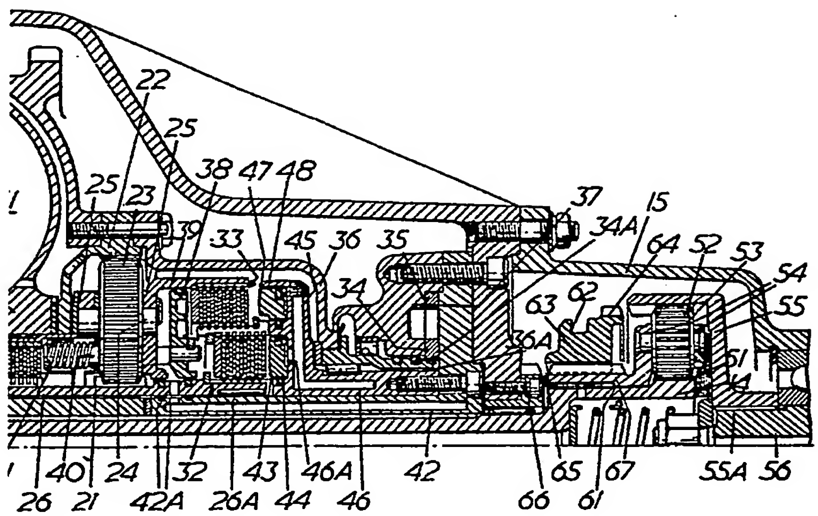
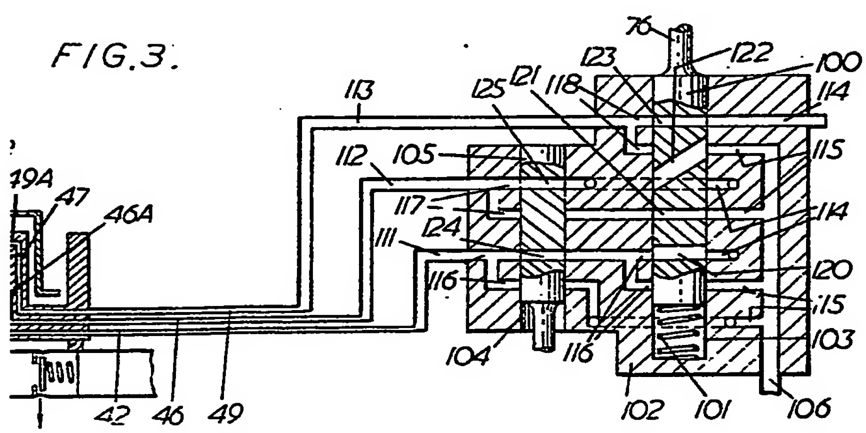


FIG. 3.



870,863 COMPLETE SPECIFICATION
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SHEET 1

FIG. 1.

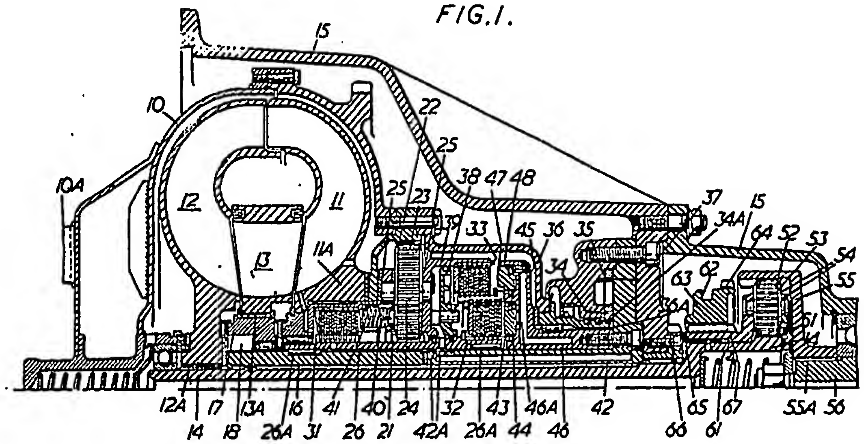


FIG. 3.

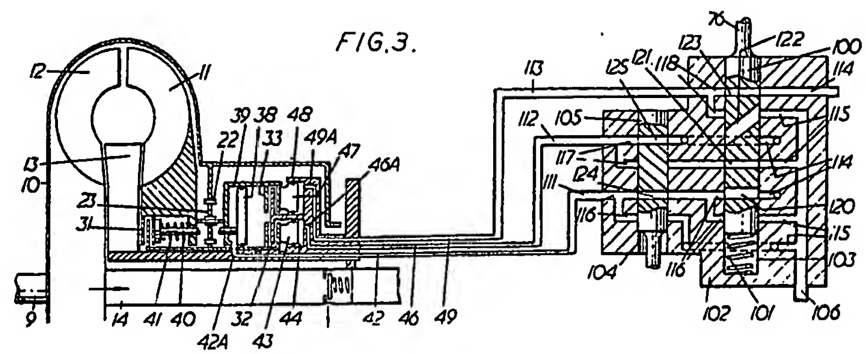
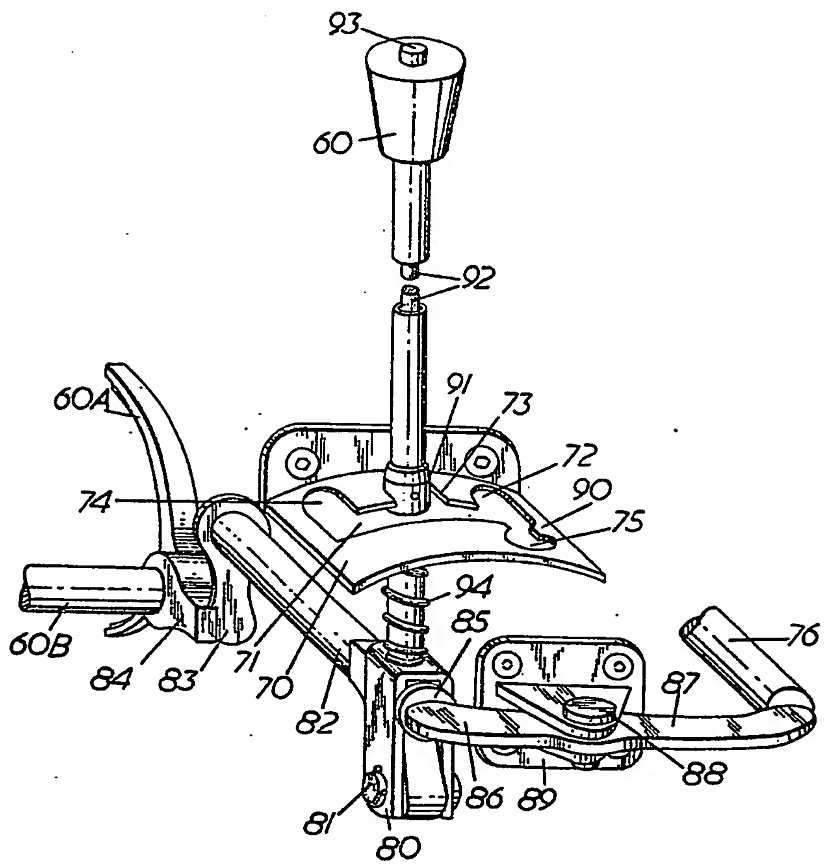


FIG. 2.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870,863 COMPLETE SPECIFICATION
3 SHEETS
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SHEETS 2 & 3

FIG.4.

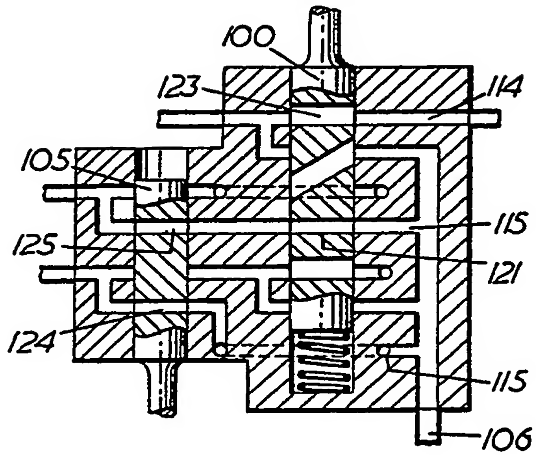

FIG.5.

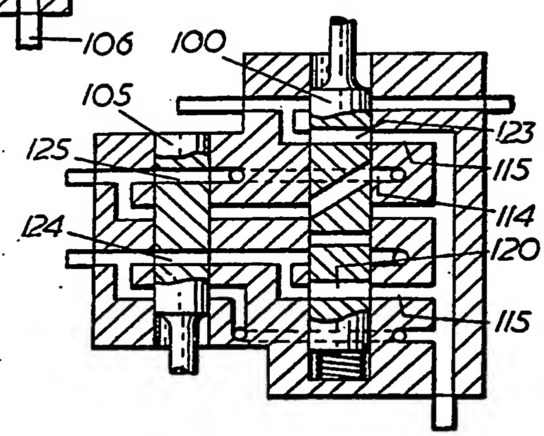

FIG.6.

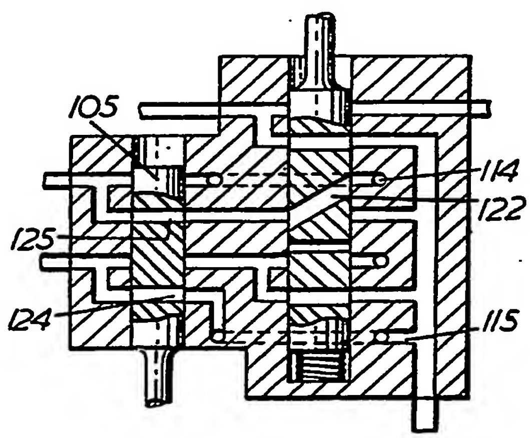

FIG.2.

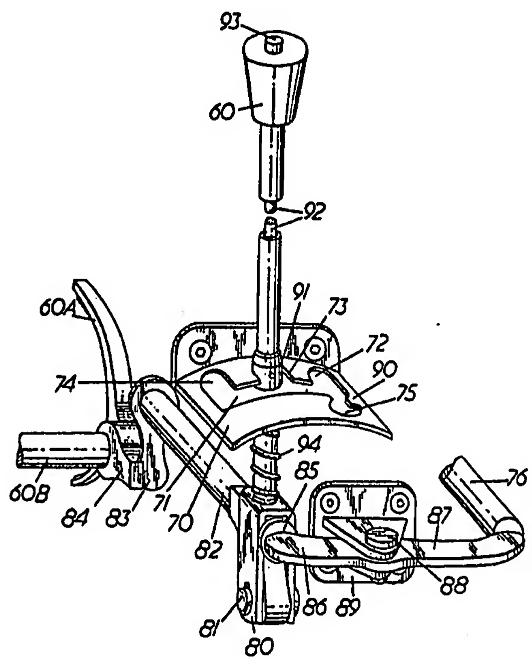

FIG.4.

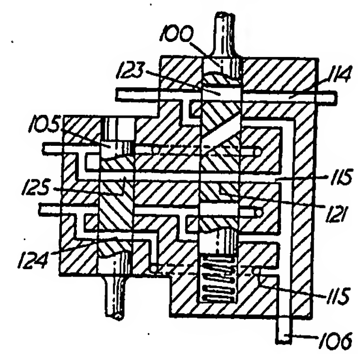

FIG.5.

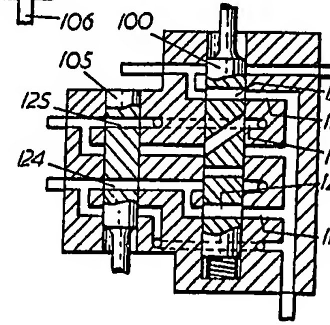

FIG.6.